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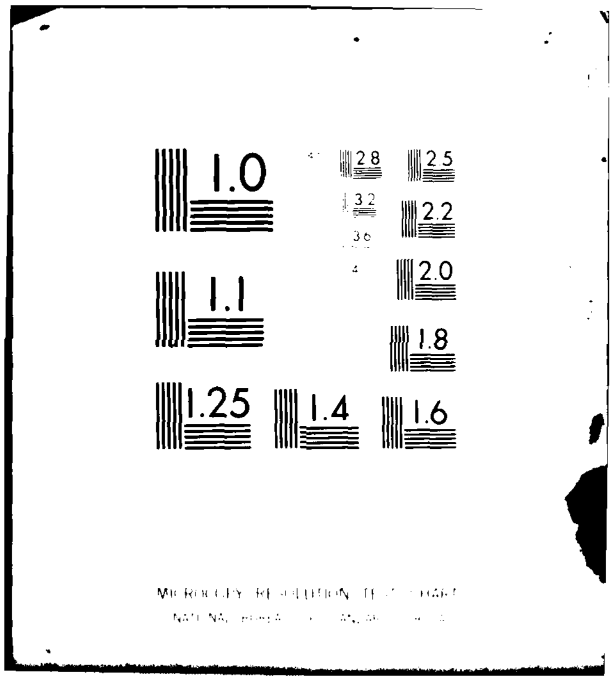
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THESIS

THE STUDY OF A ROTATING-HEAT-PIPE-COOLED
ELECTRIC MOTOR

by

Gerhard Otto Immel

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Thesis Advisor: P. J. Marto

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motor in its original configuration (i.e. with a solid shaft), with its solid shaft replaced by a rotating heat pipe, and with this heat pipe unit containing segmented external fins to increase air cooling.

The modifications did not show a decrease of motor temperatures. This was probably due to a lack of cooling air through the motor casing. Recommendations for future work are included.

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THE STUDY OF A ROTATING-HEAT-PIPE-COOLED ELECTRIC MOTOR

by

Gerhard Otto Immel
Lieutenant Commander, Federal German Navy

Submitted in partial fulfillment of the
requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

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December 1980

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ABSTRACT

The steady state temperatures of a conventional, 15 HP electric motor were compared to those of a rotating - heat - pipe - cooled motor under identical loading conditions. Fourteen thermocouples were used to measure temperatures at various locations within the motor. Seven of these were placed in rotating parts of the motor and were recorded through a mercury slip ring unit. Tests were made with the electric motor in its original configuration (i.e. with a solid shaft), with its solid shaft replaced by a rotating heat pipe, and with this heat pipe unit containing segmented external fins to increase air cooling.

The modifications did not show a decrease of motor temperatures. This was probably due to a lack of cooling air through the motor casing. Recommendations for future work are included.

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NOMENCLATURE

c_p	Specific heat (kJ/kg^{o}C)
h	Heat transfer coefficient (W/m^2 ^{o}C)
k	Thermal conductivity (W/m ^{o}C)
L	Length of a flat plate (m)
Nu	Nusselt number = $\bar{h}L/k$ (dimensionless)
Pr	Prandtl number = $\mu c_p/k$ (dimensionless)
Re	Reynolds number = Ux/ν (dimensionless)
T	Temperature (^{o}C)
U	Mean velocity of air (m/sec)

Greek symbols

δ	Thickness of laminar boundary layer (m)
ν	Kinematic viscosity (m^2/sec)
μ	Dynamic viscosity (kg/m sec)

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I. INTRODUCTION

A. BACKGROUND INFORMATION

There are 800 million motors operating in the U.S. They consume 60 percent of the electricity produced in this country, but they fail to transmit at least 10 percent of that into useful mechanical energy. This constitutes an energy loss equivalent to 200,000 bbl of oil a day.[1]

The capacity of an electric motor is usually limited by the temperature produced in its windings. To avoid burning of the windings, electric motors are divided into different insulation classes according to their ability to withstand high temperatures for a certain amount of time. The highest insulation class allows for only a steady state temperature of about 180°C. Heat dissipation plays a very important role. The lower the temperatures in the windings, the higher the motor capacity will be.

Various papers by Groll, Krähling, and Münzel [2] and Čslejšek and Polášek [3] describe tests in which a decrease in temperatures in the rotor and the stator of an electric motor could be reached by using heat pipes. The heat pipes were able to cool the motor effectively under all operating conditions up to 5,000 RPM. Fries [4] describes tests in which a decrease of rotor temperatures could be achieved by using a rotating heat pipe which was placed in the hollowed shaft of an electric motor.

B. THE ROTATING HEAT PIPE

A rotating heat pipe can be used to remove heat from a rotating system. As shown in Figure 1, it consists of three different parts:

1. evaporator section
2. condenser section, and
3. working fluid (condensate).

The working fluid transports the heat from one section to the other. During operation, the working fluid evaporates as heat is added to the pipe. The existing difference in pressure between evaporator section and condenser section drives the vapor from the evaporator to the condenser. Because the condenser section is cooled from the outside, the vapor condenses at the walls and is driven back to the evaporator section due to the hydrostratic pressure difference in the condensate. During the whole operation, the heat pipe is rotating about its longitudinal axis.

Two different kinds of rotating heat pipes have been developed: conical and cylindrical heat pipes. In conical heat pipes, the driving force which brings the condensate back from the condenser to the evaporator is produced by the centrifugal force which increases with increasing diameter of the heat pipe. In cylindrical heat pipes, there is no centrifugal acceleration to drive the condensate back to the evaporator. Instead, it is assumed that the

thickness of the condensate in the condenser decreases in the direction of the evaporator, and condensate flows due to a hydrostatic pressure gradient along the condenser length.

Although conically shaped heat pipes perform better than cylindrically shaped heat pipes [5], the recent work done by Marto and Wagenseil [5] and Weigel [6] dealt with the replacement of a conical shape by a cylindrical shape in the condenser. The high manufacturing costs for the conically shaped heat pipe was the reason for the replacement. Several types of cylindrically shaped heat pipes have been tested. These include a smooth cylinder, a spiral Noranda tube, a Hitachi Thermoexcel - C tube, a Hitachi Thermofin tube type II, and a Turbotec tube. Weigel [6] showed that the spiral Noranda tube achieved the best results. Using this kind of tube, the heat pipe was able to transfer 6.5 kW at a vapor temperature of 90°C at a rotating speed of 2800 RPM. The working fluid which was used was distilled water.

This thesis will only deal with a smooth cylinder. Although its heat transfer performance is inferior to the internally enhanced devices mentioned above, it is preferable for heat pipe applications due to the low manufacturing costs.

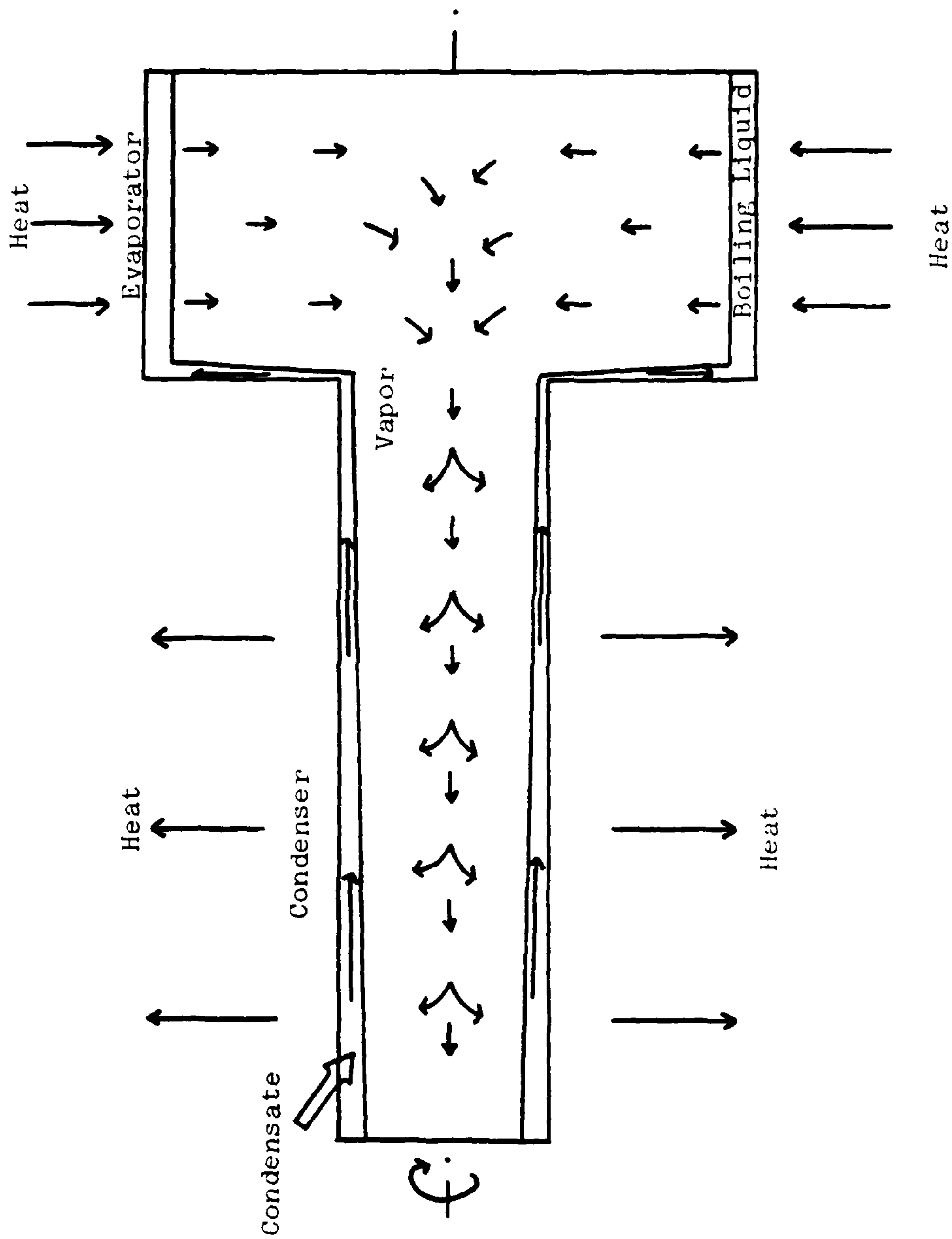

Figure 1 Working Principle of a Rotating Heat Pipe

C. EXTERNAL FINS

Because the performance of a heat pipe yields high heat transfer coefficients, the controlling resistance may be on the outside. The installation of external fins may therefore be necessary and can lead to an additional increase in the performance of the electric motor.

D. THESIS OBJECTIVE

The purpose of this thesis was therefore:

1. to compare the steady state temperatures of a conventional electric motor with the steady state temperature of a heat pipe cooled electric motor under load,
2. to design external fins which can be placed within the casing so that no additional space is necessary outside of an already existing motor, and
3. to compare the steady state temperatures of a heat pipe cooled electric motor without external fins with the steady state temperatures of the same motor with external fins.

II. EXPERIMENTAL EQUIPMENT

DESCRIPTION OF THE COMPONENTS USED DURING THE EXPERIMENTS

The following text will describe each element which was used in the experiments. (See Figures 2 and 3).

A. SYSTEM

The equipment used was a hydraulic system (manufactured by Paul Munroe, Inc.), which included an electric motor, an oil pump and reservoir, two heat exchangers, and various pressure and suction pipes. The working fluid was Chevron EP Hydraulic OIL 32. The system was designed to operate with pressure of 1,000 PSI (6.9×10^6 Pa). Several pipes were removed for the purpose of this thesis.

The oil is pumped out of the reservoir, through the pump, a pressure gage and two water-cooled heat exchangers; it then flows back into the reservoir. The pressure gage is used to monitor various load conditions.

B. ELECTRIC MOTOR (See Figure 4)

The motor driving the oil pump had a design output of 15 HP (11.18 kW) at 1175 RPM. It was a Lincoln A.C. Motor connected to 220 volts, 60 cycles with 46 amperes.

C. THERMOCOUPLES

Fourteen thermocouples were used to measure temperatures at various locations within the motor. This includes

seven thermocouples connected to rotating parts and seven thermocouples connected to stationary parts of the motor. Of the seven rotating thermocouples, three were placed at the outer surface of the rotor and four were placed along its shaft. The stationary thermocouples were placed on each of the bearings, at two positions of the stator, and at three other locations to measure inlet air, outlet air and ambient temperature. All 14 thermocouples were made of copper-constantan with teflon insulation. The rotating thermocouples were connected to a mercury slip ring unit.

D. SLIP RING UNIT (See Figure 5)

The purpose of a slip ring is to transmit measured signals from a rotating element to a stationary conductor. The slip ring unit which was used had twenty-two channels each consisting of a rotating wire, contact disc, mercury, and a stationary screw. The mercury provides an electrical conduction path without much friction or electrical noise, and consequently this type of slip ring is useful for accurate measurements. All readings of the thermocouples were displayed using a digital readout with an accuracy of $\pm 1^{o}F$.

E. SPEED CONTROL

The speed of the motor shaft was measured in RPM by a magnetic counter which was displayed on a Systron Donner Counter.

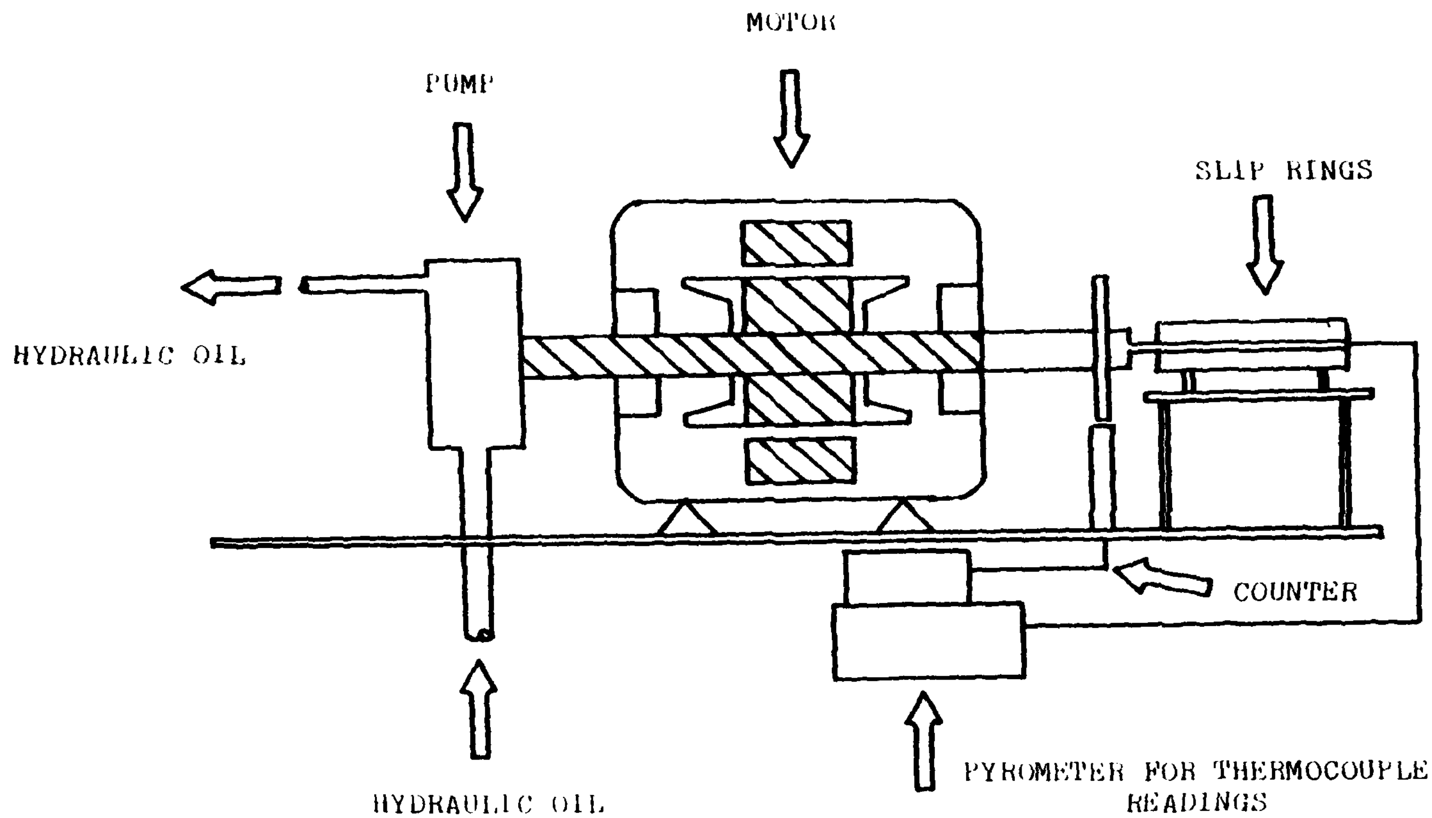

Figure 2 Experimental Set-Up

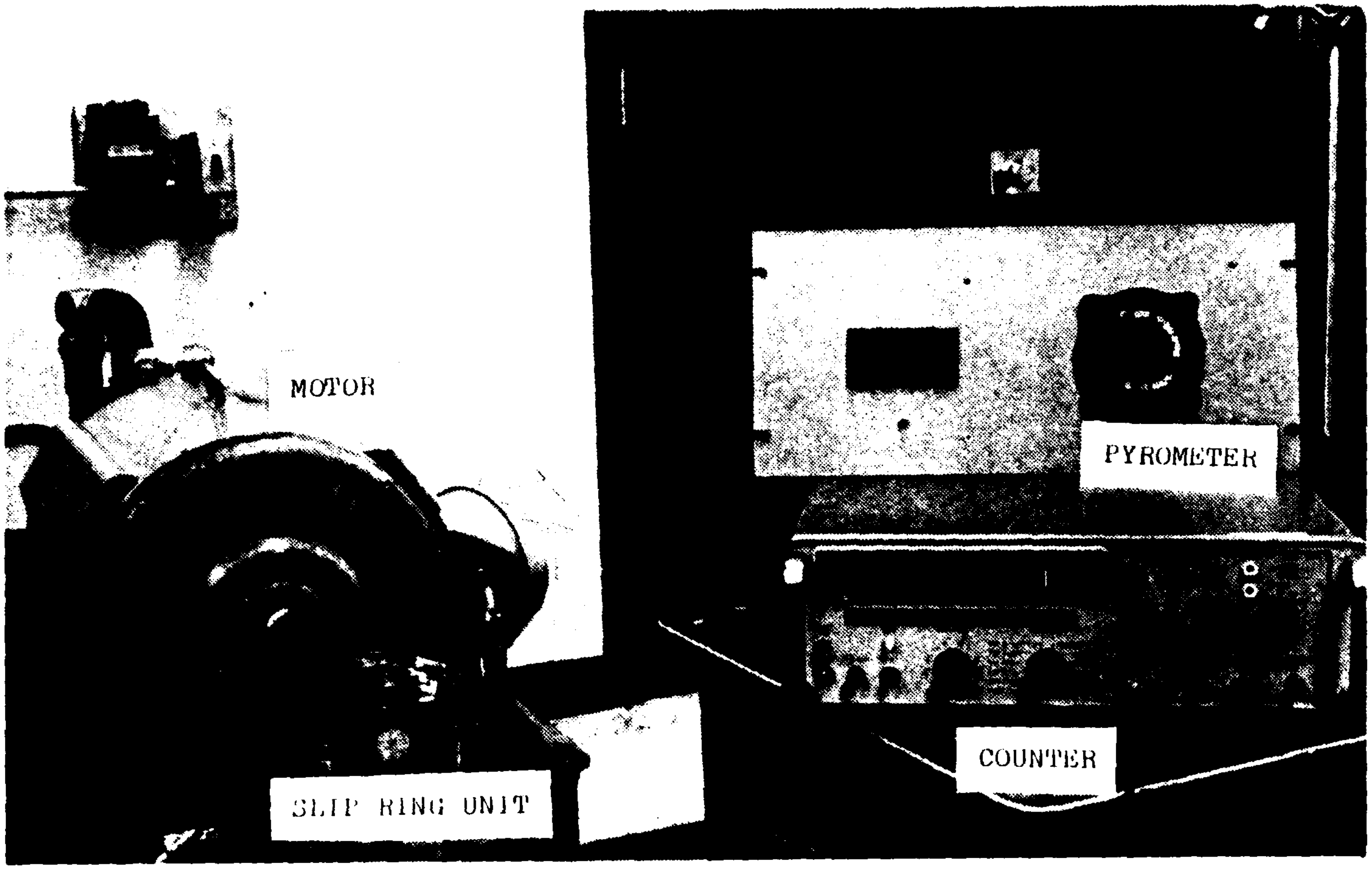

Figure 3 Photograph of the Test Facility

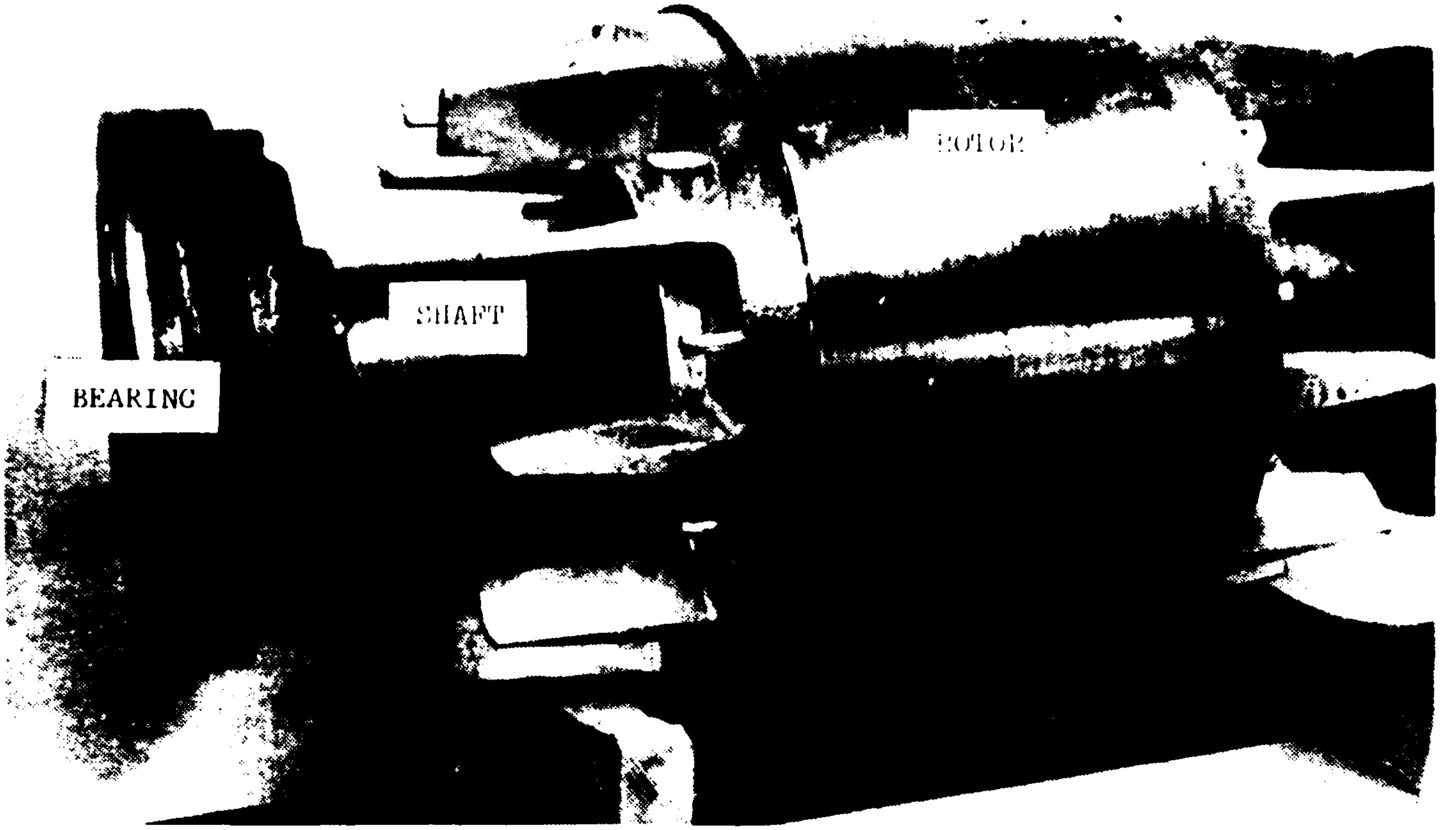

Figure 4 Photograph of the Rotor in the Configuration for the Runs with Solid Shaft

F. HEAT PIPE UNIT

The heat pipe unit was manufactured of three pieces of oxygen-free copper: a cylinder and two end caps. The cylinder consists of an evaporator and a condenser section as shown, together with dimensions, in Figure 6. The cylinder was closed and sealed at the ends with copper caps. The surfaces in the evaporator and in the condenser section were smooth. The heat pipe was designed with a step between the condenser and the evaporator of 4.7 mm. A hole with a diameter of 38.1 mm was drilled into the motor shaft, in which the described heat pipe unit was placed. To avoid bending and high stresses in the motor shaft no diameter bigger than 38.1 mm was chosen. The length of the hole was determined by the fact that the evaporator section of the unit should be surrounded by the rotor section of the motor. The condenser section of the heat pipe should be surrounded by that part of the shaft which is between the rotor and the bearing opposite to the power output side of the motor.

G. EXTERNAL CIRCUMFERENTIAL FINS

These fins were designed to dissipate the heat which was removed from the electric motor by the heat pipe. To guarantee a maximum heat dissipation, copper was chosen for the fin material. The design process is outlined in Appendix A. 104 external circumferential fins were manufactured

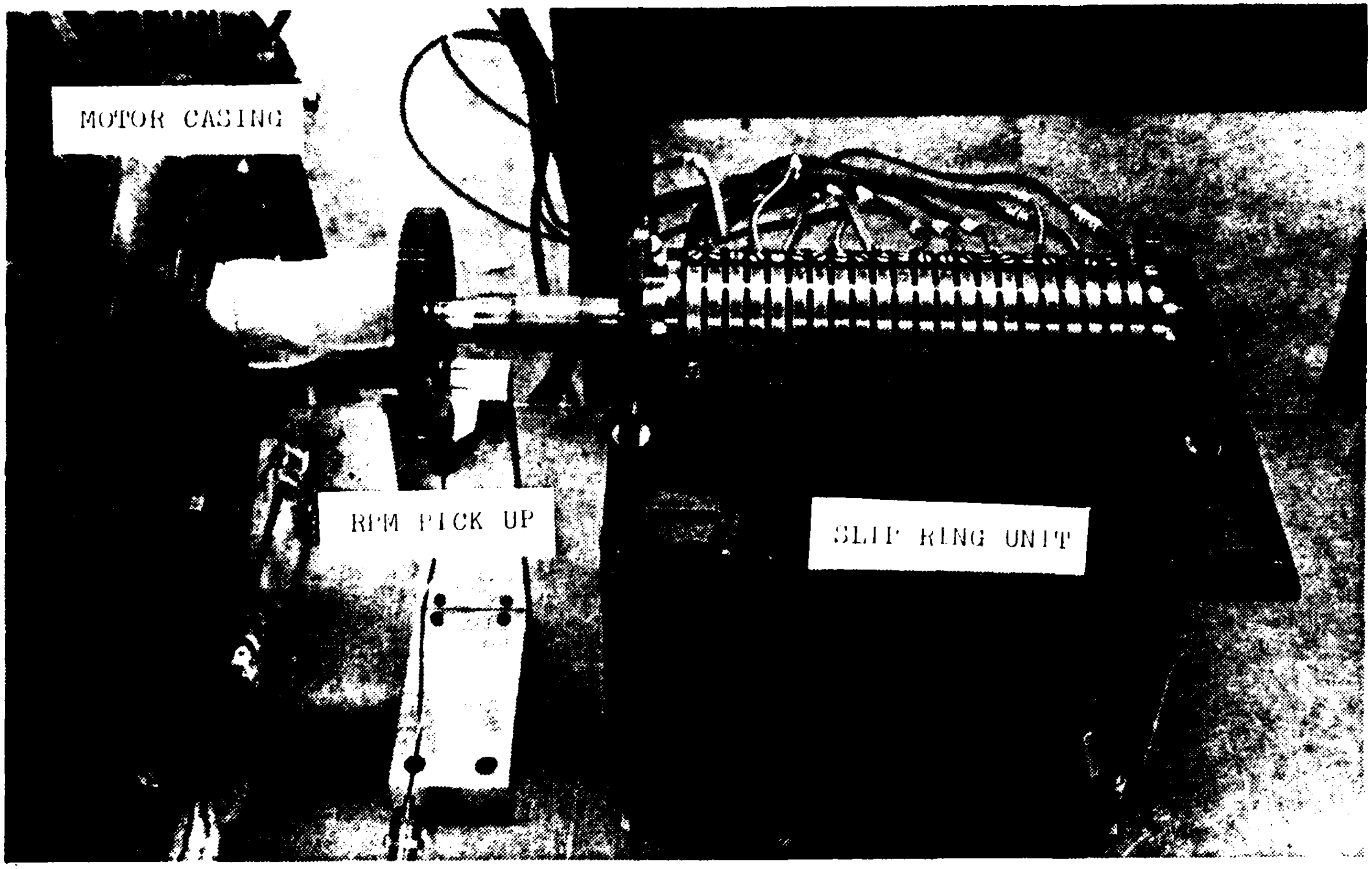

Figure 5 Photograph of the Slip Ring Unit Connected to the Electric Motor

a = 211.1 mm

b = 119.0 mm

c = 34.9 mm

d = 25.4 mm

e = 38.1 mm

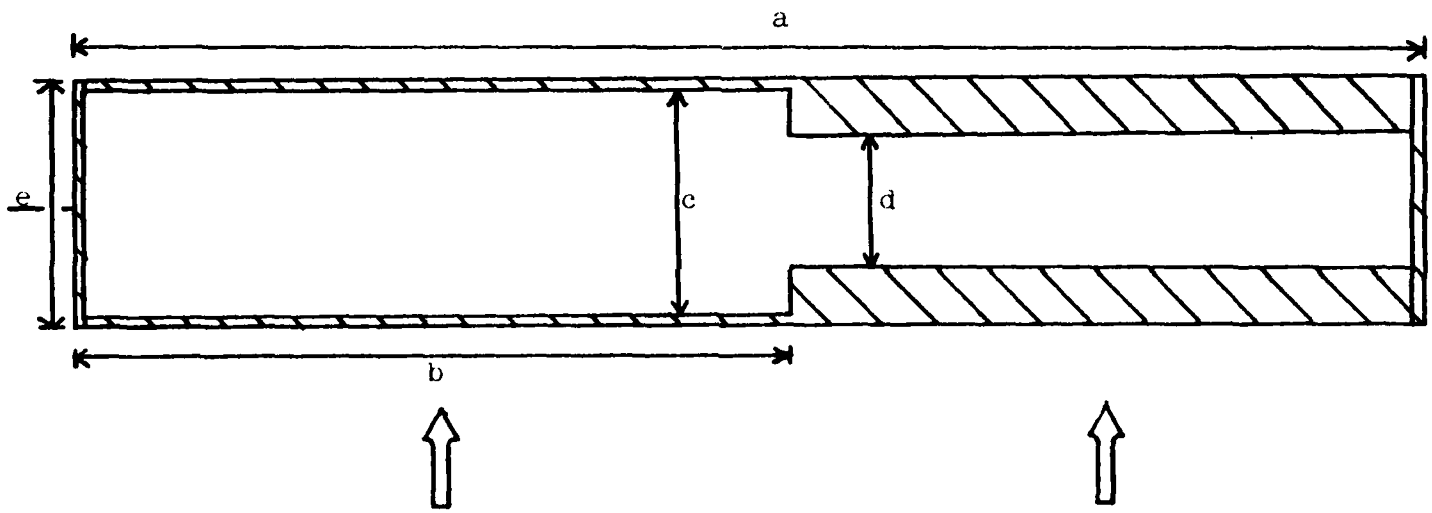

Figure 6 Heat Pipe Details

and mounted around the shaft between the rotor and the bearing opposite to the power output side. The fins were now surrounding the condenser section of the heat pipe unit. All fins had an inside diameter of 60 mm and a thickness of 0.508 mm. Twenty-six fins had an outside diameter of 130 mm, seventy-eight fins an outside diameter of 72 mm. Groups of three smaller fins were used as spacers between the bigger fins. The fins with the bigger outside diameter were cut every 45 degrees to form eight edges. These edges were bent to each side of the fin in an alternating pattern to interrupt the boundary layer. A photograph of this arrangement is shown in Figure 7.

A further modification to give better heat transfer was made by blowing air over the fins through a pipe which was supplied by an air pressure system from outside the casing.

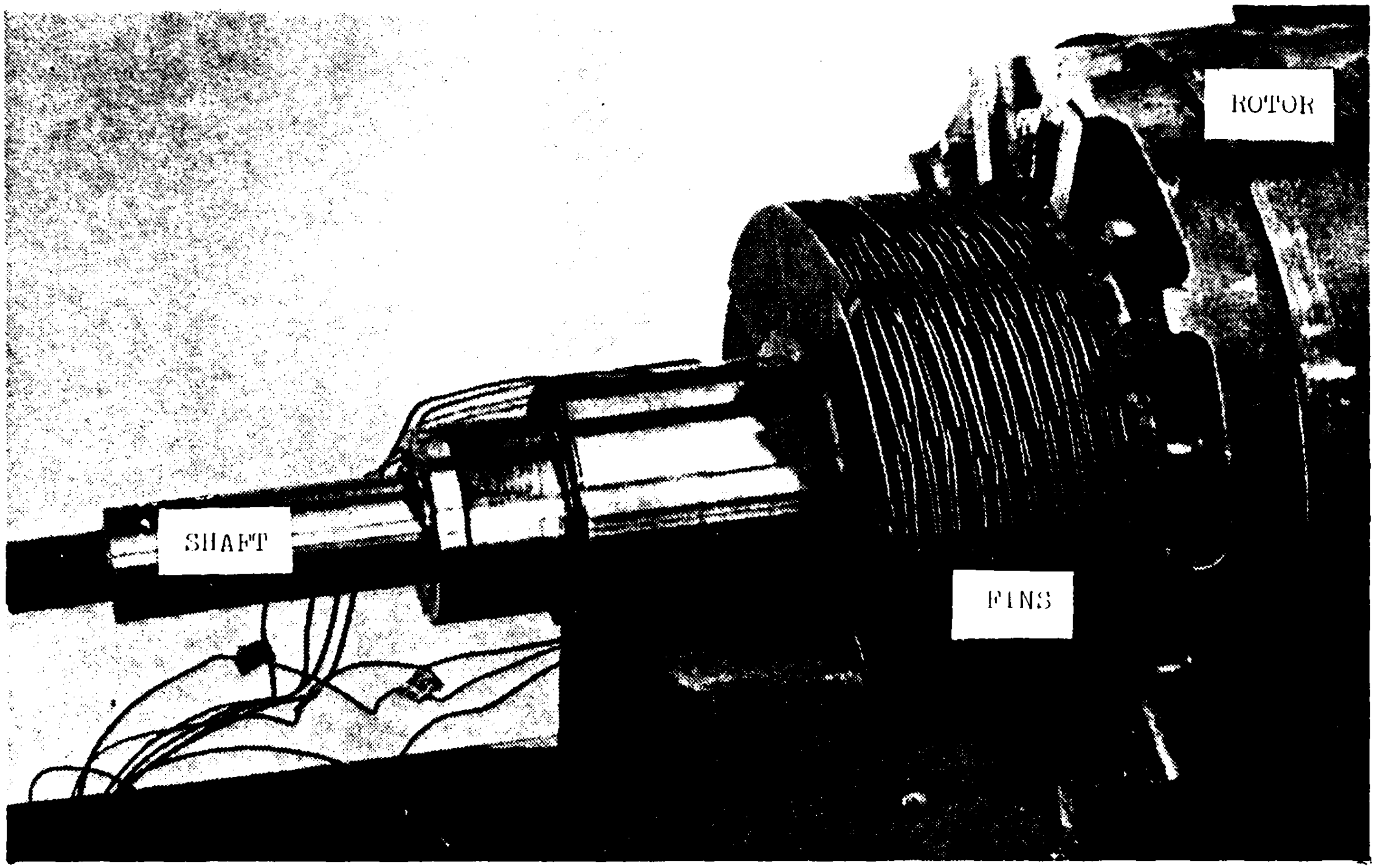

Figure 7 Photograph of the Rotor with External Segmented Fins (Without Bearing)

III. EXPERIMENTAL PROCEDURES

A. GENERAL REMARKS

The tests undertaken during this thesis work were divided into three parts:

1. The electric motor was tested in its original configuration, i.e. with a solid shaft (see Figure 8).
2. The electric motor was tested with the smooth heat pipe installed (see Figure 9).
3. The electric motor was tested with the smooth heat pipe installed together with external segmented fins (see Figure 10).

Each of the above described parts were again divided into three different operating tests involving three different loading conditions:

1. The motor was tested with a hydraulic system pressure of 600 PSI (= 60% of the design pressure) up to a point where steady state temperatures were reached.
2. The motor was tested with a hydraulic system pressure of 1,000 PSI (= 100% of the design pressure) up to steady state temperature.
3. The motor was tested with a hydraulic system pressure of 1,200 PSI (= 20% overload over system pressure). This loading condition was operated for only one hour to protect the pump and the motor.

B. TESTING PROCEDURES

During operation, the temperatures were measured every five minutes until steady state conditions were reached or after one hour was over, respectively.

1. Preparation of the Condenser Wall

To assure filmwise condensation on the surface of the condenser, it was treated with a chemical substance consisting of equal parts of a 50% solution of sodium hydroxide (NaOH) and ethanol (C_2H_5OH) warmed to 80°C. The inside of the condenser was scrubbed with a brush. As an indication of filmwise condensation, the surface was well wetted with water. After this treatment, both ends of the heat pipe were closed and sealed with copper caps.

2. Evacuation and Filling Procedure

The configuration shown in Figure 11 was used to evacuate the system and to fill the heat pipe unit with the working fluid. The following filling procedure was applied:

1. The distilled water reservoir (2) was filled with working fluid.
2. The valve (5) was opened to fill the heat pipe unit with the desired amount of working fluid. Valve (4) was closed during this operation.
3. Valve (5) was closed while valve (4) was opened to evacuate the system to the desired vacuum level. At the same time, the heat pipe unit was heated for degassing the working fluid.

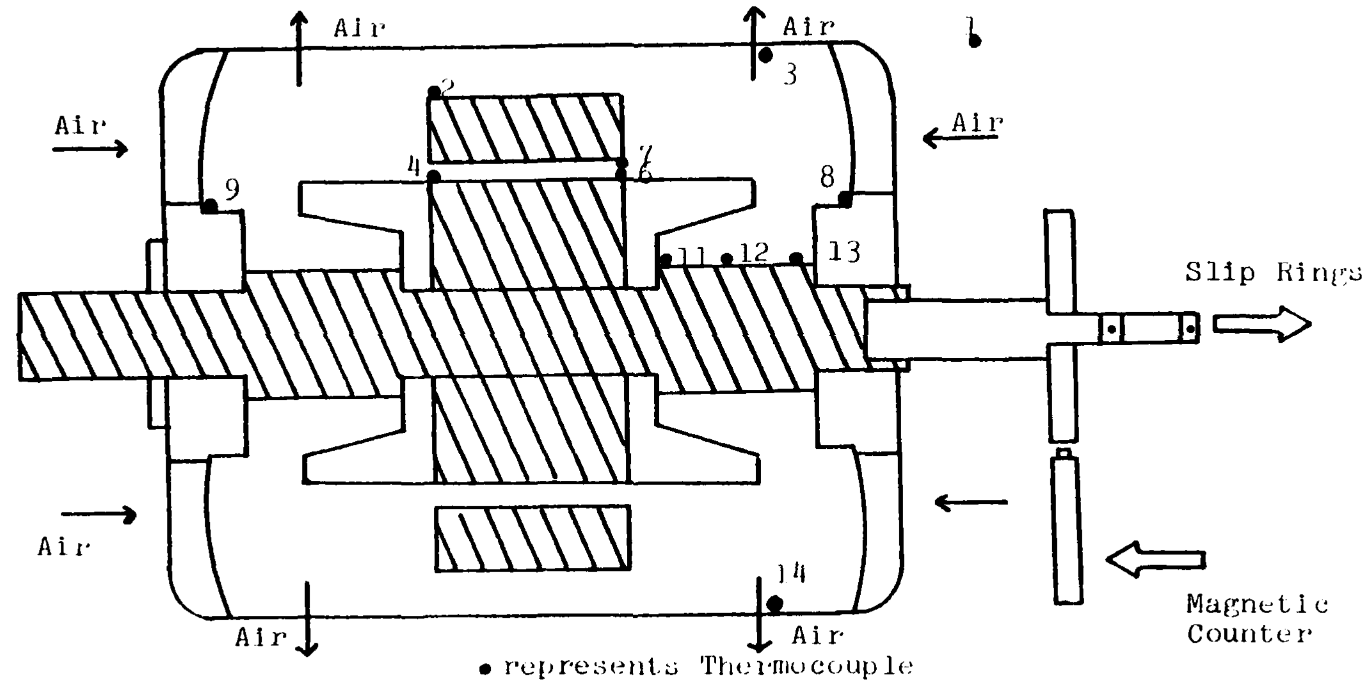

Figure 8 Configuration of the Electric Motor During the Runs With the Solid Shaft

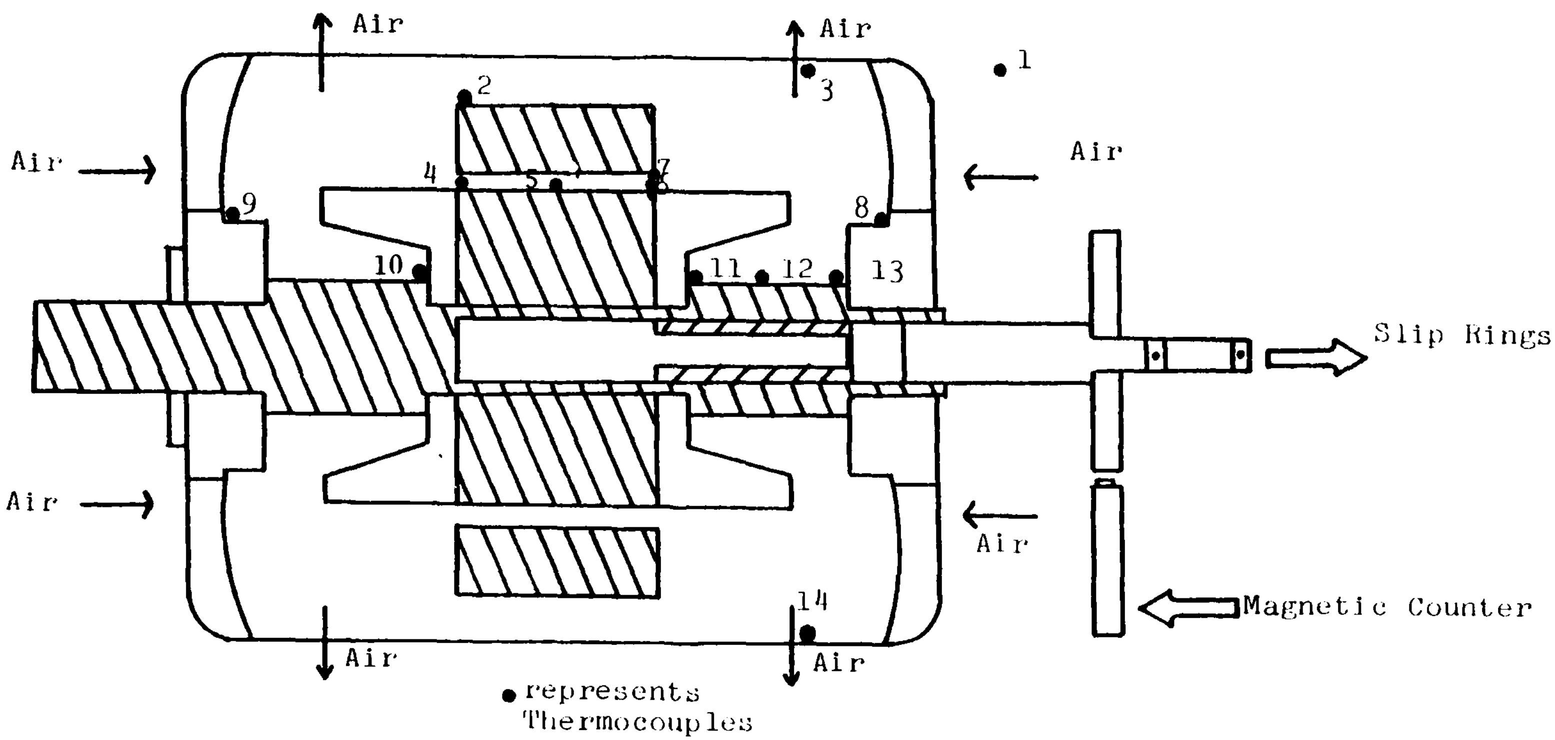

Figure 9 Configuration of the Electric Motor During the Runs With the Heat Pipe Installed

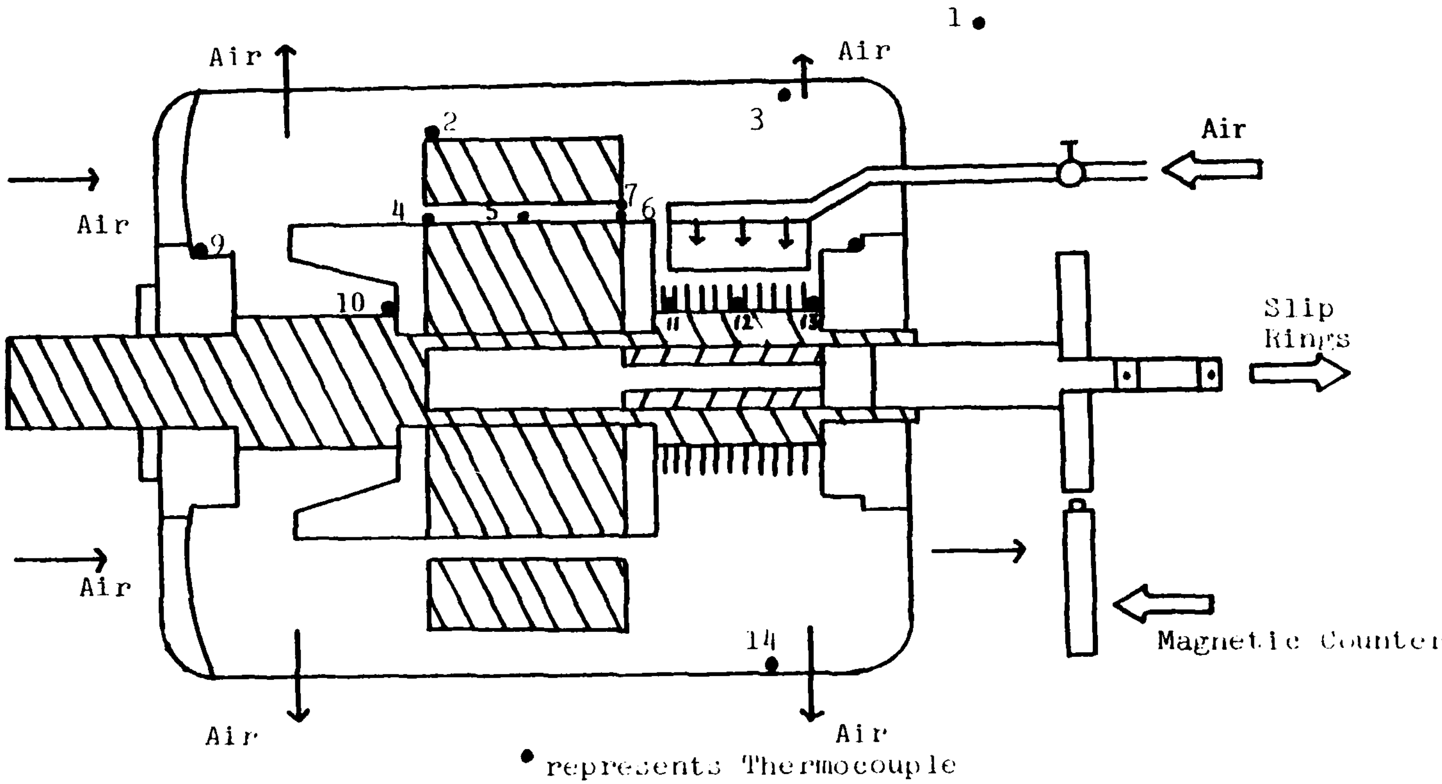

Figure 10 Configuration of the Electric Motor During the Runs with the Heat Pipe Installed Together with External Segmented Fins

4. To assure that the vacuum could be maintained, valve (4) was closed and the vacuum was observed with gage (6).

5. When it was certain that the heat pipe was vacuum tight, the unit was sealed by crimping at point 7.

The volume of working fluid added was 45.5 ml. After filling the system, the working vaccum was 98.2 kPa (29 in Hg).

After evacuation and filling, the heat pipe unit was shrink-fitted into the shaft by using a press. The heat pipe unit was cooled down to -20°C and the shaft was heated with a gun before the two were pressed together.

1 - heat pipe unit
2 - distilled water reservoir
3 - high vacuum pump
4 - valve
5 - valve
6 - vacuum gage

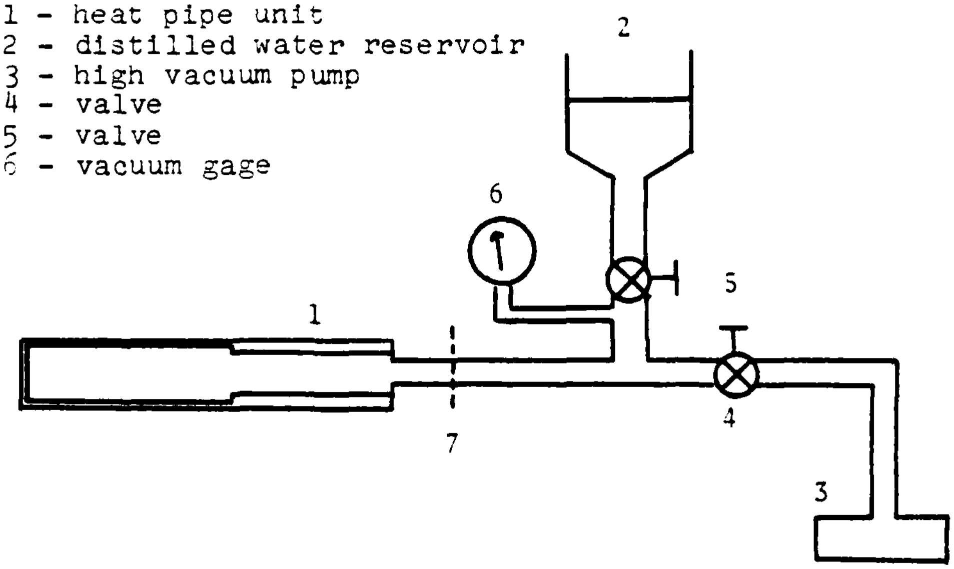

Figure 11 Apparatus Used for Evacuation and Filling of the Heat Pipe Unit

C. OPERATION

During all tests in this thesis the following working procedure was used:

1. Open the cooling water inlet into the two heat exchangers and adjust the flow rate to about 23 liters per minute (6.1 gallons per minute). Because the two heat exchangers were in parallel, each heat exchanger received about 50% of the flow rate.
2. Take thermocouple readings.
3. Start the electric motor.
4. Select the desired working pressure of the hydraulic system.
5. Take thermocouple readings, pressure reading, and speed reading every five minutes. Also check flow rate.
6. Repeat step 5 until steady state temperatures are reached or, in the case of the loading of 1,200 PSI, until one hour is over.

Load \ Run	1	2	3
600	22	21	17
1000	22	20	17
1200	26	22	17

Thermocouple 1

Load \ Run	1	2	3
600	41	42	38
1000	50	49	47
1200	53	55	49

Thermocouple 2

ALL TEMPERATURES IN ^{o}C

Table 1 Temperature Readings for Thermocouples 1 and 2

Run / Load	1	2	3
600	30	31	39
1000	35	35	47
1200	37	36	48

Thermocouple 3

Run / Load	1	2	3
600	45	46	46
1000	57	58	59
1200	63	64	62

Thermocouple 4

ALL TEMPERATURES IN ^{o}C

Table II Temperature Readings for Thermocouples 3 and 4

Load \ Run	1	2	3
600	-	48	49
1000	-	61	65
1200	-	67	66

Thermocouple 5

Load \ Run	1	2	3
600	43	46	48
1000	54	55	60
1200	58	60	63

Thermocouple 6

ALL TEMPERATURES IN $^{\circ}C$

Table III Temperature Readings for Thermocouples 5 and 6

Load \ Run	1	2	3
600	43	44	47
1000	56	56	61
1200	60	61	64

Thermocouple 7

Load \ Run	1	2	3
600	27	22	42
1000	30	22	50
1200	32	22	49

Thermocouple 8

ALL TEMPERATURES IN ^{o}C

TABLE IV Temperature Readings for Thermocouples 7 and 8

Load \ Run	1	2	3
600	31	37	27
1000	34	40	32
1200	34	41	32

Thermocouple 9

Load \ Run	1	2	3
600	-	42	41
1000	-	48	51
1200	-	52	52

Thermocouple 10

ALL TEMPERATURES IN ^{o}C

TABLE V Temperature Readings for Thermocouples 9 and 10

Load \ Run	1	2	3
600	36	43	42
1000	45	52	52
1200	47	54	53

Thermocouple 11

Load \ Run	1	2	3
600	34	42	41
1000	42	50	51
12000	43	52	52

Thermocouple 12

ALL TEMPERATURES IN ^{O}C

TABLE VI Temperature Readings for Thermocouples 11 and 12

Load \ Run	1	2	3
600	33	43	41
1000	39	50	51
1200	42	52	52

Thermocouple 13

Load \ Run	1	2	3
600	30	31	40
1000	35	35	49
1200	36	36	50

Thermocouple 14

ALL TEMPERATURES IN ^{o}C

TABLE VII Temperature Readings for Thermocouples 13 and 14

IV. PRESENTATION AND DISCUSSION OF RESULTS

A. RESULTS OF RUN 1 - MOTOR WITH ORIGINAL SOLID SHAFT

With increasing loading conditions, the temperature at all locations increased. The measured temperature distribution was as expected. Relatively high temperatures were observed in the windings of the stator (thermocouple 2) and between stator and rotor (thermocouples 4, 6, and 7). Lower temperatures could be seen along the shaft. The difference between the air inlet temperature and outlet temperature (thermocouples 1 and 3) was 8°C for the load at 600 PSI, 13°C for the load at 1,000 PSI, and 11°C for the load at 1,200 PSI. (Note: Load of 1,2000 PSI does not include steady state temperatures!). Run 1 was mainly to provide temperature readings for comparison to run 2 and run 3.

B. RESULTS OF RUN 2- MOTOR WITH ROTATING HEAT PIPE SHAFT (NO EXTERNAL FINS)

Increasing loading conditions produced higher temperatures in all locations of the motor. The highest temperatures for each load were found in the windings (thermocouple 2) and between stator and rotor (thermocouples 4, 5, 6, and 7). Along the shaft, the temperatures increased (thermocouples 11, 12, and 13) up to 12°C in comparison with run 1 at all loads. This increase in temperature was an indication that the heat pipe was operating properly.

However no decrease of temperature in other locations of the motor could be noticed in comparison with all loads of run 1. The difference between the air inlet temperature and outlet temperature (thermocouples 1 and 3) was 10°C for the load at 600 PSI, 15°C for the load at 1,000 PSI, and 14°C for the load at 1,200 PSI.

C. RESULTS OF RUN 3 - MOTOR WITH ROTATING HEAT PIPE SHAFT (WITH EXTERNAL FINS)

Again, increasing loading conditions led to higher temperatures at all locations of the motor. The highest temperatures were found in the windings (thermocouple 2) and between the rotor and the stator (thermocouples 4,5,6 and 7). However, the temperatures measured by thermocouples 4, 5, 6, and 7 were now 5°C higher than in run 1. Temperatures measured with thermocouples 3, 8, and 14 were up to 17°C higher in comparison to the run with the solid shaft. The difference between air inlet temperature and outlet temperature (thermocouples 1 and 3) was 22°C for the load at 600 PSI, 30°C for the load at 1,000 PSI, and 31°C for the load at 1,200 PSI.

After cutting the original rotor fins at the heat pipe side of the motor apparently only a very small amount of air was blown through that part. The air which was provided by an external air supply was blown only over the added segmented fins and not through the whole casing. If it is

assumed that the efficiency of the motor used in this thesis was 90%, the heat losses of the motor would be 1.5 HP or 1118 Watts. With the assumption that the heat losses in the rotor are approximately 50% of 1118 Watts or 559 Watts, it is obvious that the designed fins with an ability to dissipate a maximum of 100 Watts (see Appendix A) were not able to cool the rotor effectively. Even if the heat transfer of the fins could be increased to about 200 Watts by segmenting them, there is still a lack of 360 Watts which could not be dissipated.

V. CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

Based upon the experimental results the following conclusions can be made:

1. The designed heat pipe was able to remove the motor heat along the shaft. The temperatures measured by thermocouples 11, 12, and 13 increased for the runs with the installed heat pipe in comparison to the runs with the solid shaft.
2. Although the heat pipe was able to bring heat to the shaft, it was not able to cool the rotor effectively. The reason for this may be the large thermal resistance of the steel within the rotor and a poor air flow and air path design in the original configuration of the motor.
3. After adding the external segmented fins around the shaft, which surrounds the condenser part of the heat pipe, the temperatures at this end of the motor increased rather than decreased due to a reduced air flow in the casing.
4. For better results, the condenser part of the rotating heat pipe should be placed outside of the casing of the motor. A better cooling of the fins would then be possible.

B. RECOMMENDATIONS

Research in the future might be done in the following areas:

1. The use of a generator instead of a hydraulic system may be preferrable to load the motor. It would be possible to overload the motor and to produce unusually high temperatures.
2. Instead of designing circumferential fins within the motor casing, different external fin arrangements outside the casing should be tested. This would include an extension of the motor shaft. As a consequence, the dimensions of the heat pipe unit would be increased. The evaporator section would lie within the casing and the condenser section would be placed in the extended section of the shaft surrounded by the fin arrangement. This arrangement would allow liquid, or spray cooling of the heat pipe fins.
3. Different kinds of surfaces in the condenser section should be tested (e.g. a spiral Noranda tube) to increase the ability of the heat pipe unit to dissipate more produced heat.
4. This research should be coordinated with representatives of the Department of Electrical Engineering in order to properly select motor conditions for heat pipe evaluations in the future.

5. The heat pipe should be instrumented with a thermocouple in the evaporator section to obtain the temperature distribution inside the heat pipe.

APPENDIX A

THEORETICAL PROCEDURES

A. DESIGNING OF EXTERNAL CIRCUMFERENTIAL FINS

For the purpose of reducing the controlling resistance on the outside of the heat pipe, external circumferential fins were designed. The space between the rotor and the bearing opposite to the power output side of the motor was chosen as a proper place for the external fins. A finite difference method taught by Yovanovich [7] was used to design the dimensions, the shape, and the number of circumferential fins, which were necessary to dissipate the heat produced by the motor.

For the designing process, some simplifications and assumptions were made:

1. The thickness of the fins are constant throughout their entire length (rectangular cross section of constant thickness).
2. The base temperature of the fins and the spacers between the fins is uniform.
3. The thermal conductivity (k) of the fin material is assumed to be constant throughout the fin length.
4. The heat transfer coefficient (h) of air is uniform along the fin length. It was calculated assuming laminar flow over a flat plate.

Therefore [8],

$$Nu_L = 2\ Nu_{x=L}$$

where:

$$Nu_{x=L} = \bar{h}L/k = 0.332\ Re_L^{1/2}\ Pr^{1/3} \qquad \text{(A-1)}$$

where:

Nu = Nusselt number hL/k (dimensionless)

h = heat transfer coefficient ($W/m^2 {}^oC$)

L = length of a flat plate

k = thermal conductivity (W/m^oC)

Pr = Prandtl number = $\mu c_p/k$ (dimensionless)

Re = Reynolds number = Ux/ν (dimensionless)

5. It was assumed that the flow between each pair of fins is Blasius flow. The thickness of the boundary layer can be calculated by [9] .

$$\delta = 5\ (\nu L/U)^{1/2}$$

where:

ν = kinematic viscosity

L = length of a flat plate

U = mean velocity of air

It was assumed that the minimum distance between the fins should be twice the thickness of the laminar boundary layer at the end of the effective length L.

To calculate the heat transfer coefficient of air the following model was used:

1. Based upon the given velocity of the shaft of 1175 RPM, the velocity in meters per second at a point A halfway between

the inside radius and the outside radius b (which is assumed twice the inside radius a) could be calculated. This was evaluated as 5.3 meters per second.

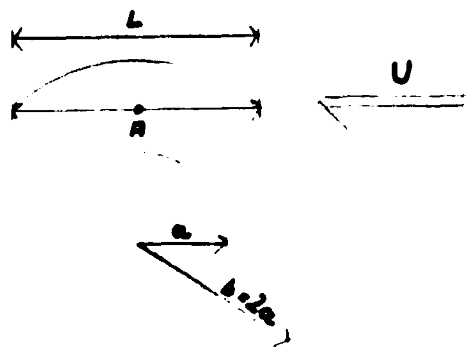

Figure 12 Circumferential Fin Details

2. With the assumption that the maximum temperature of the air which will flow over the fin is 60°C, the Reynolds number can be calculated:

$$Re = \frac{UL}{\nu} \qquad (A-3)$$

where:

L = length of the path of air through point A

ν = kinematic viscosity of air at 60°C

U = velocity of air past point A

The Reynolds number was calculated as 2×10^4. Because the Reynolds number was below 5×10^5, it was assumed that the flow over the fin is laminar.

3. Knowing the Prandtl number, the heat transfer coefficient of air could be evaluated as 32.13 $\frac{W}{m^2C}$ (5.65 $\frac{Btu}{hrft^2F}$)

4. To calculate the amount of heat transferred from the inside of the heat pipe, through the copper wall of the heat pipe, through the steel wall of the shaft, to the base of the fin the one-dimensional steady state conduction method for a cylinder was used. The maximum amount of heat transferred from the base of one fin to the surrounding air was calculated by using a finite difference method taught by Yovanovich [7]. With the space available, a choice was made between fin thickness, fin spacing and fin length in order to calculate the maximum heat which could be transferred. Assuming the thermal conductivity of steel as 40 $\frac{W}{mC}$, the thermal conductivity of copper as 350 $\frac{W}{mC}$, the heat transfer coefficient within the heat pipe at 70°C (as maximum temperature in the heat pipe) as 1420 $\frac{W}{m^2C}$, the following parameters could be calculated.

a. Number of fins: 104

b. Maximum heat transfer: 100 Watts

Due to the fact that the space along the shaft between the rotor and the bearing was very limited (about 60 mm), and as a consequence a relatively small amount of heat could be dissipated, the circumferential fins were replaced by segmented fins to interrupt the boundary layer and enhance the heat transfer. This increase in heat transfer

was expected based upon results presented by Mori and Nakayama [10]. The interruption of the boundary layer is a very important concept to increase the heat transfer coefficient.

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